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Alternative approaches to thermosyphon solar-energy water heater performance analysis and characterisation

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Abstract

Simplified models, performance correlations and rigorous simulation models are identified as alternative techniques for the prediction and characterisation of the performance of natural-circulation solar energy water heaters. The features of particular examples of each approach are described. In their different appropriate contexts, each technique is shown to provide good agreement with measured system behaviour. A more general role in practical system design is foreseen for rigorous detailed simulation. Models have been used primarily for generic research optimisations. © 2000 Elsevier Science Ltd. All rights reserved.

1. Introduction

The heat transfer fluid circulates from the solar-energy collector to the hot-water store in a thermosyphon solar-energy water heater by buoyancy forces. Neither a pump nor its associated control sensors and actuators are required to circulate the heat-transfer fluid. Such solar water heaters thus have both lower initial and running costs compared with equivalent forced-circulation units. Furthermore, thermosyphon solar-energy water heaters can perform as effectively as forced-circulation units [4].

The performance of natural-circulation solar-energy water heaters may be predicted analytically by three broad alternative approaches:

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Nomenclature

A	area (m^2)
C	mean specific heat capacity (J kg^{-1} K^{-1})
D	diameter of the collector flow channel (m)
F	collector's efficiency factor
f	solar fraction
H	total daily insolation upon collector (Jm^{-2})
h	height (m)
I	global insolation falling upon the plane of collector (Wm^{-2})
L	length of flow channel (m)
M	mass of water (kg)
$\dot{m}$	mean daily thermosyphonic mass flow rate (kg s^{-1})
N	number of times the water contained in the store is circulated through the collector during the insolation period
n	integer
$\dot{Q}$	mean rate of delivery of useful heat from the collector during the insolation period (W)
Q	total heat delivery (J)
s	collector-plane's inclination to the horizontal (Degrees)
T	temperature (K)
t	time (s)
U	steady-state heat loss co-efficient (Wm^{-2} K^{-1})
X, Y Z	dimensionless groups defined by Eqs. (26), (27) and (28), respectively
α	absorptance of the absorber surface
β	coefficient defined by Eq. (32)
ΔT	temperature difference between the fluid at the inlet and outlet of the collector (K)
η	efficiency
θ	duration of insolation period (s)
τ	transmittance of collector glazing
ϕ	$T_s - T_a$

Subscripts

a	of the ambient environment
c	for the collector
d	refers to the downcomer pipe
e	effective
f	for the heat-transfer fluid
i	refers to collector inlet
L	refers to water withdrawn from store

m	mean value
n	refers to number of passes
p	for the collector plate
R	heat-removal factor
s	refers to the system as a whole
T	for the hot-water store
u	refers to the upriser pipe
z	for the volume of water to be heated
1	refers to the fully-mixed store, immediately prior to the insolation period

Superscripts

′	denotes absorber plate efficiency factor
-	denotes average value of the variable

- simplified models, providing average or cumulative diurnal predictions
- correlation of performance characteristics from either simulation or monitoring of generic systems, and
- rigorous dynamic simulation models using detailed numerical analysis, providing hour-by-hour or more frequent performance predictions

2. Simplified models

A multiple-pass solar water heater is defined as a system in which the volume of water in the store is circulated through the collector many times each day. It has been observed in numerous experimental studies that, for a multiple pass thermosyphon solar-energy water heater, the difference in the fluid temperatures at the inlet and outlet of the solar-energy collector is constant [5]. Whether a particular system is single or multiple pass depends on the system's geometry and flow resistance as well as on the prevailing insolation level. An almost constant temperature differential across a collector (concomitant with multiple-pass behaviour) ensues, provided either a critical height difference between the collector and the elevated hot-water store for a fixed rate of heat input, or a critical insolation level for a fixed system geometry has been exceeded [6].

The exhibition of an almost constant temperature differential across the collector has been used to develop a particular simplified analysis — the 'ΔT model' — for the determination of daily system performance [5]. The advantage of this particular approach is that there is no necessity to determine the frictional flow losses within the fluid circuit nor to perform a time-wise analysis of the temperature variation around the thermosyphon loop to ascertain the buoyancy pressure head.

To develop any simplified model, many assumptions are required. For the 'ΔT model', these are:

1. the temperature differential, ΔT, across the collector remains constant throughout the insolation period. The duration of periods at the start and end of each insolation period during which the system does not exhibit this behaviour are assumed to be much shorter than the insolation period.
2. diurnal mean insolation and ambient temperatures can be employed in the analysis without introducing significant errors in the calculated values of the net daily heat gain.
3. at the start of the insolation period, the hot-water store is fully mixed at a temperature T_I. During the period of insolation, the mean store temperature rises linearly with time to a value T_z at sunset. No hot water is withdrawn from the system during this period. The latter assumption implies that the system is intended to produce hot-water for withdrawal either that evening or early the subsequent day.
4. all fluid properties are independent of temperature.

For no mixing in the hot-water store, the water temperature in the store after n passes of the full volume of the store through the collector is:

$$T_n = T_1 + n\,\Delta T \tag{1}$$

This is also the inlet temperature, $T_{f,i}$ for the (n+1) pass. If the total number of passes is N, then at the end of the day

$$T_z = T_1 + N\,\Delta T \tag{2}$$

$$\bar{T}_{f,i} = \left(\frac{1}{N}\right)\sum_{n=1}^{N} T_{f,i} = T_1 + \left(\frac{N}{2}\right)\Delta T \tag{3}$$

$$= T_1 + \frac{(T_z - T_1)}{2} \tag{4}$$

Alternatively, when the store is fully mixed, the rate of increase of the store temperature is:

$$\frac{\mathrm{d}T}{\mathrm{d}t} = \Delta T\left(\frac{\dot{m}}{M}\right) \tag{5}$$

thus

$$T(t) = T_1 + \Delta T + t\left(\frac{\dot{m}}{M}\right) \tag{6}$$

At the end of insolation period,

$$T_z = T_1 + \Delta T\,\theta\left(\frac{\dot{m}}{M}\right) \tag{7}$$

The inlet temperature at the collector is assumed synonymous with the tank temperature, throughout the insolation period. The diurnal average fluid inlet temperature at the collector can be found by taking the diurnal average of the tank temperature, i.e.,

$$\bar{T}_{f,i} = T_1 + \left(\frac{1}{2}\right)\Delta T\cdot\theta\left(\frac{m}{M}\right) \tag{8}$$

$$= T_1 + \frac{(T_z - T_1)}{2} \tag{9}$$

Thus the mean fluid temperature over the whole day at the inlet to the collector is given by Eq. (4), irrespective of the degree of mixing in the store. The useful rate of heat provided by a solar energy collector can be described by:

$$\dot{Q} = F_{\mathrm{R}}A_{\mathrm{c}}[\bar{I}\tau\alpha - U_{\mathrm{c}}(\bar{T}_{f,i} - T_a)] \tag{10}$$

where F_{R}, the heat removal factor is given by

$$F_{\mathrm{R}} = \frac{mc_f}{A_{\mathrm{c}}U_{\mathrm{c}}}\left(1 - \exp\left(\frac{-U_{\mathrm{c}}A_{\mathrm{c}}F'}{\dot{m}c_f}\right)\right) \tag{11}$$

The rate of energy delivery by the collector is:

$$\dot{Q} = \dot{m}c_f\Delta T \tag{12}$$

Substituting from Eqs. (4), (10) and (11) into Eq. (12) gives:

$$\frac{A_{\mathrm{c}}}{m} = \frac{-c_f}{U_{\mathrm{c}}F'}\log_{\mathrm{e}}\left\{1 - \frac{U_{\mathrm{c}}\Delta T}{I\tau\alpha - U\left[\left(\frac{T_1 + T_z}{2}\right)\right] - T_{\mathrm{a}}}\right\} \tag{13}$$

From the energy balance for the whole system, over the full insolation period the average collector plate temperature is:

$$\bar{T}_{\mathrm{P}} = F_{\mathrm{R}}\left[\left(\frac{1}{F_{\mathrm{R}}} - 1\right)\left(\frac{\bar{I}\tau\alpha}{u} + \bar{T}_{\mathrm{a}}\right) + \left(\frac{T_1 + T_z}{2}\right)\right] \tag{14}$$

for which it is assumed that the heat losses from the up-riser and down-comer are negligible, thus:

$$T_z = \frac{T_1\left(M_z C_f - \frac{U_T A_T \theta}{2}\right) + \dot{m}\Delta T C_f \theta}{m_z C_f - \frac{U_T A_T \theta}{2}} + \bar{T}_a U_T A_T \theta \tag{15}$$

The system's diurnal heat-gain efficiency is defined as

$$\pi_s = \frac{M_z C_f (T_z - T_1)}{\bar{I}\theta A_c} \tag{16}$$

The critical minimum vertical height of the down-comer which will prevent circulation occurring at night is given by [5].

$$h_d = \frac{1}{(T_u - T_d)}\{(T_d - T_c) l_c \sin s + (T_z - T_u) h_T \tag{17}$$

If, for the particular installation, the assumption that, at night,

$$T_c = T_a, \quad T_d = T_a + 5 \text{ (in°C) and } T_u = \frac{T_z + T_a}{2}$$

adequately reflect reality, then Eq. (17) becomes

$$h_d = \frac{1}{\left\{\frac{T_z - T_a}{2} - 5\right\}}\left\{5 l_c \sin s + \frac{(T_z - T_a)}{2} h_T\right\} \tag{18}$$

Using initial estimates for T_z and T_P, U_c is calculated. Using Eq. (14) a new value of $\bar{T}_p$ is calculated, this process is repeated until convergence. A new value for T_z is found from Eq. (15) and, using the convergence value of $\bar{T}_p$, an iterative procedure is used to obtain a solution for T_z. If mutually-incompatible parameters are introduced into the procedure, then no solution will be found. This is because Eq. (13) cannot be solved as:

$$U_c \Delta T > \bar{I}\tau\alpha - U_c\left\{\frac{(T_1 + T_z)}{2} - T_a\right\} \tag{19}$$

When compatible values for the overall collector heat loss co-efficient are chosen, the condition expressed in Eq. (19) will not arise; for example if the value of U_c is large then a small ΔT would be expected.

Theoretical predictions from the ΔT model have been compared with measured performance data from two identical thermosyphonic solar-energy water heaters [6]. Each 1.4 m^2 collector consisted of single-glazed aluminium flat plate absorbers, painted matt black: their fourteen flow channels were of 25 mm internal diameter. They were inclined at 60° to the horizontal and were oriented facing south. The

storage tanks were each 1.6 m tall and 0.3 m in diameter, giving an aspect ratio of 5.3 and capacity of 0.1125 m^3. In each system the upriser discharged into the tank at a distance of 30 mm below its top, and the downcomer was connected at the level of 30 mm above the tank base. The data acquired included the time histories of temperatures at various heights in the collector store and the ambient temperature using copper/constantan thermojunctions. The global insolation on the collector surface was measured with a Moll-Gorczynski pyranometer, mounted in the same plane as the collectors. During the tests, no water was withdrawn from the store. Comparisons of the predicted and observed values for the mean temperature of the stored hot water are shown in Fig. 1. These results, obtained for a wide range of ambient and initial system conditions, show good agreement. The model could not be applied to the system for those days during the test period with a final mean store temperature of $<38°C$, as the collector temperature differential did not exhibit a constant value throughout the majority of the insolation period.

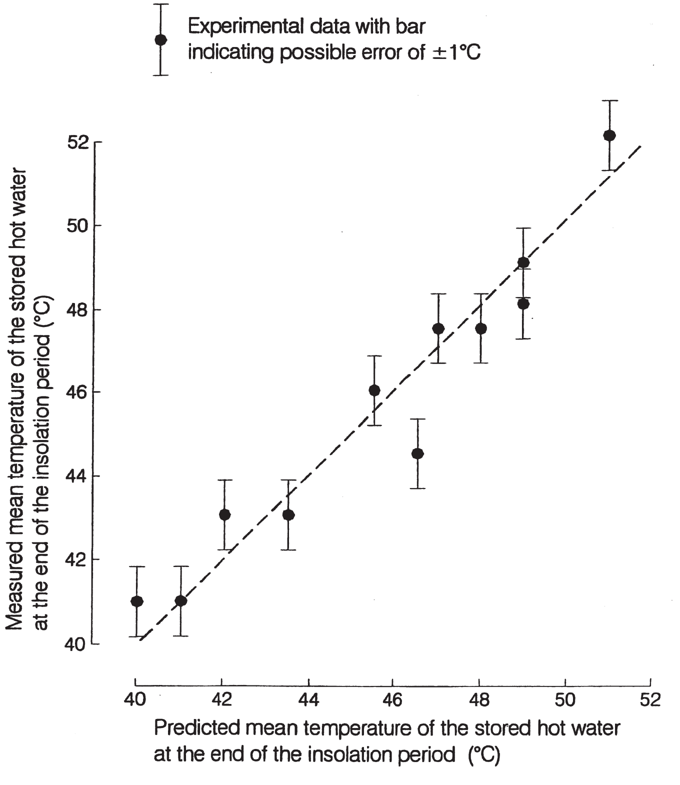

Fig. 1. Comparison of the experimentally-observed final mean temperature of the stored hot water with that predicted by the 'ΔT' simplified model. No water withdrawal from store.

3. Correlation of performance characteristics

In this approach, using simplifying assumptions, an analysis of the thermal behaviour is undertaken to identify dimensionless groups. The developed correlations between these characterising groups describe long-term system performance. As the simplifying assumptions do not apply in all cases, the functional relationship between the dimensionless correlating parameters obtained analytically is not that observed in reality. The actual correlation between the identified grouped parameters is obtained either by comparison with the output of a rigorous numerical simulation model or data from long-term system performance monitoring.

In a particular approach to the correlation of the daily performance of indirect thermosyphon solar-energy water heaters [2], a transient heat balance was undertaken on a generic solar-energy water heater (see Fig. 2), with the following assumptions:

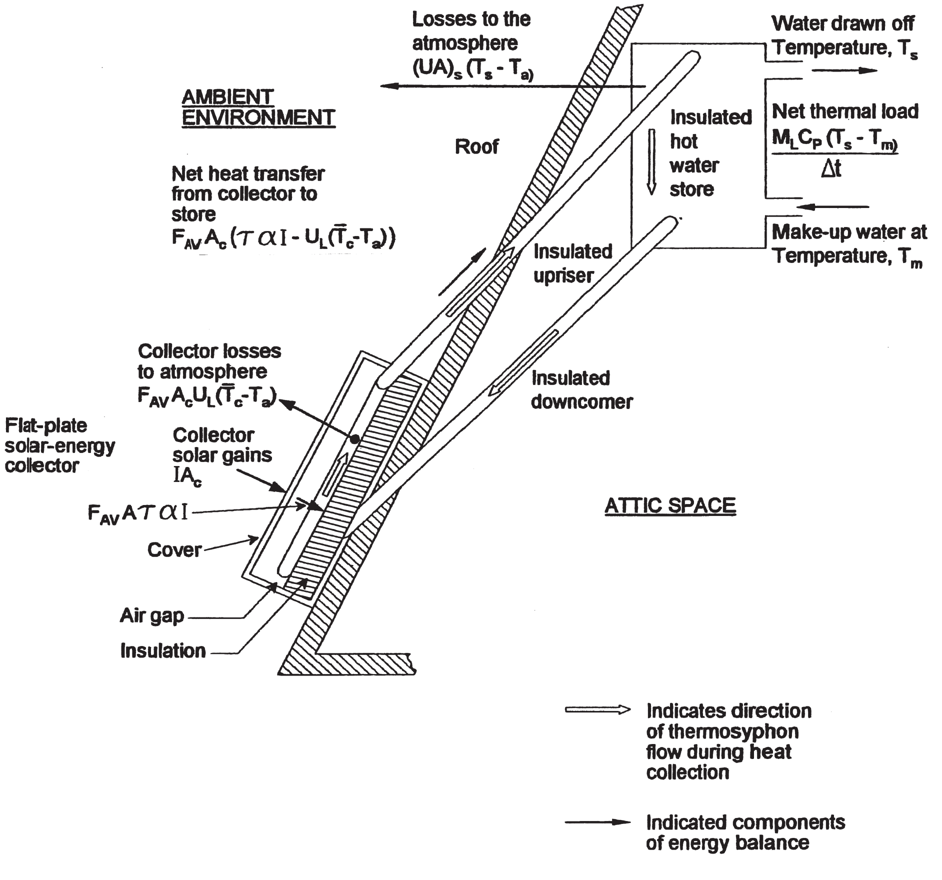

Fig. 2. Holistic heat transfer mechanisms for a generic thermosyphon solar energy water heater.

(i) the collector and storage tank both have the same mean diurnal temperatures,
(ii) insolation and ambient temperature are assumed to remain invariant over the day at their respective mean measured values,
(iii) the total draw-off of water from the store takes place as a single event at the end of the insolation period,
(iv) water is drawn off at the mean temperature of the store and replenished at the mains water supply temperature,
(v) tank and pipe losses are negligible, and
(vi) the store is initially at the mains water supply temperature at the onset of the insolation period.

An instantaneous heat balance on the system gives, with assumption (i)

$$M_T C_p \frac{\mathrm{d}T_T}{\mathrm{d}t} = \bar{F}_R A_c [(\tau\alpha)_e \frac{H}{\theta} - U(T_T - T_a)] \tag{20}$$

Using assumptions (ii) and (v), after making the substitution and rearranging, Eq. (20) becomes,

$$\frac{\mathrm{d}\phi}{\mathrm{d}t} + \frac{\bar{F}_R A_c U \phi}{M_s C \rho} = \frac{\bar{F}_R A_c (\tau\alpha)_e H}{M_s C_p \theta} \tag{21}$$

With assumption (vi) as a boundary condition, the solution to Eq. (21) is

$$\phi - \phi_o = \left[\frac{\bar{F}_R A_c (\tau\alpha)_e H - \bar{F}_R A_c U \theta (T_m - T_a)}{\bar{F}_c A_c U_L \theta}\right]\left[1 - \exp\left(\frac{\bar{F}_R A_c U}{M_T C_p}\right)\right] \tag{22}$$

The difference in the mean store temperature between the beginning and the end of the insolation period is therefore

$$\phi - \phi_o = \left[\frac{\bar{F}_R A_c (\tau\alpha)_e H - \bar{F}_R A_c U \theta (T_m - T_a)}{\bar{F}_c A_c U \theta}\right]\left[1 - \exp\left(\frac{\bar{F}_R A_c U_L \theta}{M_s C_p}\right)\right] \tag{23}$$

From assumptions (iii) and (iv), the solar fraction can be expressed as:

$$f = \frac{M_L C_p (\phi - \phi_1)}{Q} \tag{24}$$

Substituting Eq. (23) into (24) gives:

$$\frac{fQ}{M_L C_p (T_m - T_a)\left[1 - e - \left(\frac{\bar{F}_{RVA_c} U_\theta}{M_T C}\right)\right]} = \frac{\bar{F}_R A_c (\tau\alpha)_e H}{\bar{F}_R A_c U \theta (T_m - T_a)} + 1 \tag{25}$$

from which three dimensionless groups can be defined as:

$$X=\frac{fQ}{M_{\mathrm{L}}C(T_{\mathrm{m}}-T_{a})}\ \frac{\text{total heat delivered by system}}{\text{change in internal energy of water drawn-off when raised from mains to ambient temperature}} \tag{26}$$

$$Y=\frac{\bar{F}_{\mathrm{R}}A_{\mathrm{c}}(\tau\alpha)_{\mathrm{e}}H}{M_{\mathrm{L}}C(T_{\mathrm{m}}-T_{a})}=\frac{\text{total daily insolation absorbed}}{\text{change in internal energy of store when raised from mains to ambient temperature}} \tag{27}$$

$$Z=\frac{\bar{F}_{\mathrm{A}}A_{\mathrm{c}}U\theta}{M_{\mathrm{L}}C}=\frac{\text{total collector heat loss coefficient}}{\text{heat capacity of store}} \tag{28}$$

So in terms of the dimensionless groups defined, Eq. (25) becomes

$$\frac{X}{1-\mathrm{e}^{-z}}=\frac{Y}{Z}+1 \tag{29}$$

Eq. (29) therefore indicates a linear relationship between $\frac{X}{1-\mathrm{e}^{-z}}$ and $\frac{Y}{Z}$. The effectiveness of the particular correlation method described has been tested via comparison with data for two hundred and fifty complete days automated monitoring [7] of an indirect thermosyphon solar-energy water heater retrofitted to a family dwelling. The values of the mean ambient and mains water supply temperatures, total insolation and mass of solar heated water drawn off from the store were calculated from the available data in order to determine the daily values of the dimensionless groups X, Y and Z defined by Eqs. (26), (27) and (28), respectively. The daily solar fraction, f, was determined from

$$f=\frac{\Sigma[M_{\mathrm{L}}C(\bar{T}_{T}-T_{m})]}{Q} \tag{30}$$

For the particular thermosyphon solar-energy water heater described, inspection of the dimensionless groups X, Y and Z indicates a linear relationship, shown graphically in Fig. 3, between the grouped parameters is applicable to $\bar{H}(\tau\alpha)$, $\bar{F}_R$, U_L, A_c and M_T which defined the system configuration and thermal characteristics. The performances of buoyancy-driven systems also depend on other factors such as the position of the store relative to the collector. The influence of these other factors may be encapsulated in the relationship between the store and collector temperatures, that is the validity, or otherwise of assumption (i). An initial investigation indicated a constant temperature difference between the daily mean collector (T_c) and store (T_T) temperature over the insolation period. Fig. 4 shows a plot of the daily values for T_c against T_S for the month of August indicating a linear relationship. The best straight line fit through the data gave:

$$T_c - 1.00\ T_T = -1.41 \tag{31}$$

with a correlation coefficient of 0.97. Rearranging Eq. (31) gives:

$$T_T - T_c = 1.41$$

Thus, more generally

$$T_T - T_c = \beta \tag{32}$$

Introducing into the analysis such a constant temperature difference β, between the store and collector results in the dimensionless group Y being redefined as,

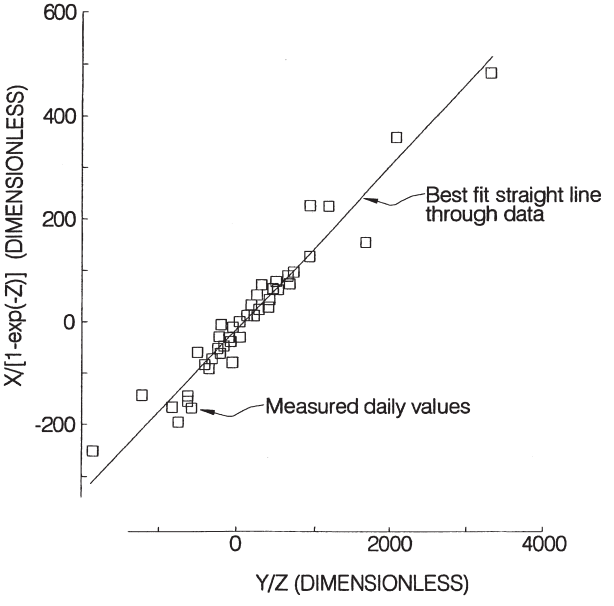

Fig. 3. $\frac{X}{(1-\exp(-z))}$ and $\frac{Y}{Z}$ correlated over a period of 250 days.

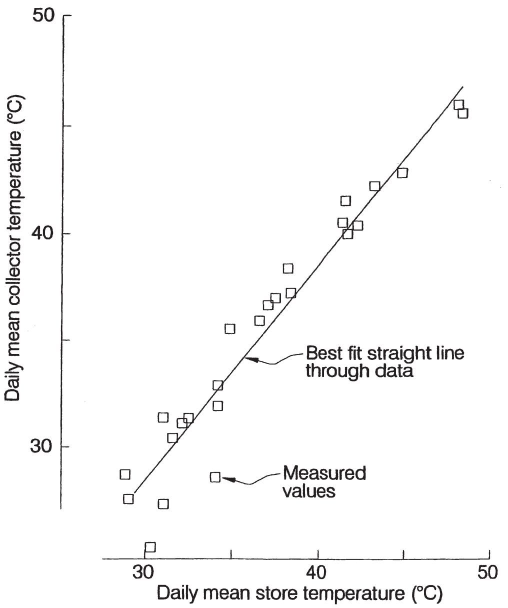

Fig. 4. Correlation of the mean collector and hot water store temperatures, each averaged over single periods of insolation.

$$Y=\frac{F_{\mathrm{AV}}A_{\mathrm{c}}[(\tau\alpha)_{\mathrm{e}}H+U_{\mathrm{L}}\beta\theta]}{M_{T}C_{\mathrm{p}}(T_{\mathrm{m}}-T_{a})} \tag{33}$$

Using this modified value of Y results in a correlation with a more universal application potentially. The performance of any system can then be determined from several daily measurements of β. Experimentally-determined values of β would enable X, Z and a modified Y parameter to be used to predict the daily performance of the system under the applied conditions with a universal correlation. Alternatively, daily values of β could be generated using a high-level rigorous simulation model and correlated against the height of the store relative to that of the collector. The latter approach is employed by [7] to develop a single sizing-chart nomogram.

4. Rigorous dynamic simulation models

In a rigorous model, the coupled energy and momentum equations which describe the transient behaviour of a natural-circulation solar-energy water heater are solved numerically. The main features of a representative example of a dynamic simulation model [3] are:

(i) a finite difference, transient heat transfer analysis is applied to the liquid circul-

ating through all the components (i.e., collector, upriser, hot-water store and downcomer) of the thermosyphon loop,
(ii) the thermal capacitances of both the collector plate and the cover are included,
(iii) the density, specific heat, viscosity, conductivity and Prandtl number of the circulating liquid are represented as second-order polynomial functions of temperature,
(iv) all the heat-transfer coefficients are temperature dependent and based on the ambient and mean component temperatures and are updated at each time step in the numerical calculation,
(v) the store model includes buoyancy-induced mixing between stratified layers which occurs when warmer fluid is introduced below a cooler layer,
(vi) a transient momentum equation is employed,
(vii) friction factors are calculated using correlations appropriate to both the non-isothermal thermally-destabilised low Reynolds number flow and isothermal developing laminar flow in straight pipes,
(viii) empirically determined laminar heat-loss coefficients for the pipe bends are included,
(ix) time variations of insolation, ambient temperatures and hot-water withdrawal encountered by an installed system during operation [8], were used as inputs to the simulation,
(x) the transmission of the glass collector cover is determined as a function of the sun hour angle, and
(xi) a two-dimensional finite-difference model for the flat-plate solar-energy collector was employed. The main advantages of this over previously used procedures are
(a) thermal masses and temperature profiles for each of the fluid, collector plate and glass cover can be described individually; and
(b) fewer simplifying assumptions are made about both the heat transfer processes within the collector plate and between the collector plate and the fluid.

The equations are expressed in finite difference form and solved simultaneously using the appropriate boundary conditions. In this model an implicit method of solution was employed in which N simultaneous equations are set up for the N nodes. and equations were solved using a Gauss–Seidel iterative method. For this method, the solution is stable unconditionally and the size of the time step is only limited by the required accuracy of the prediction of performance.

Validation was undertaken in an indoor thermosyphonic solar-energy water heater test apparatus, the measured global insolation was simulated experimentally by regulating, via a computer, the electrical power supplied to the heater. Similarly, the thermal load on the system was reproduced experimentally by the controlled opening and closing of the solenoid valves giving a draw-off pattern equivalent to that produced by the occupants of the house. The measured insolation, draw-off pattern and ambient and mains temperatures were used as input data to the performance simulation. Measured values of the global insolation and water draw-off patterns were obtained from an occupied dwelling in the south-east of England to which a thermo-

syphon solar-energy water-heater had been installed and monitored [7]. Fig. 5 shows the variation of the experimental and predicted mean store temperatures during the day. The predicted mean store temperatures agree to within 0.7°C of the corresponding measured values over the period of time from 00.00 hours to approximately 20. 10 hours, when a 12 kg water draw-off was made. After this draw-off, the predicted mean store temperature fell to 1.3°C below the measured value, i.e. the predicted temperature of the water drawn off from the top of the store was higher than the actual temperature of the draw-off water.

Good agreement is shown, as illustrated in Fig. 6, between the measured and predicted values for both the collector's inlet and outlet temperatures over the duration of the simulated insolation period. Discrepancies between these sets of values, particularly at the end of the insolation period, are due predominantly to the uncertainty in predicting the frictional flow losses at the very low Reynolds number flows encountered during reverse circulation. The predicted flow rates shown in Fig. 7

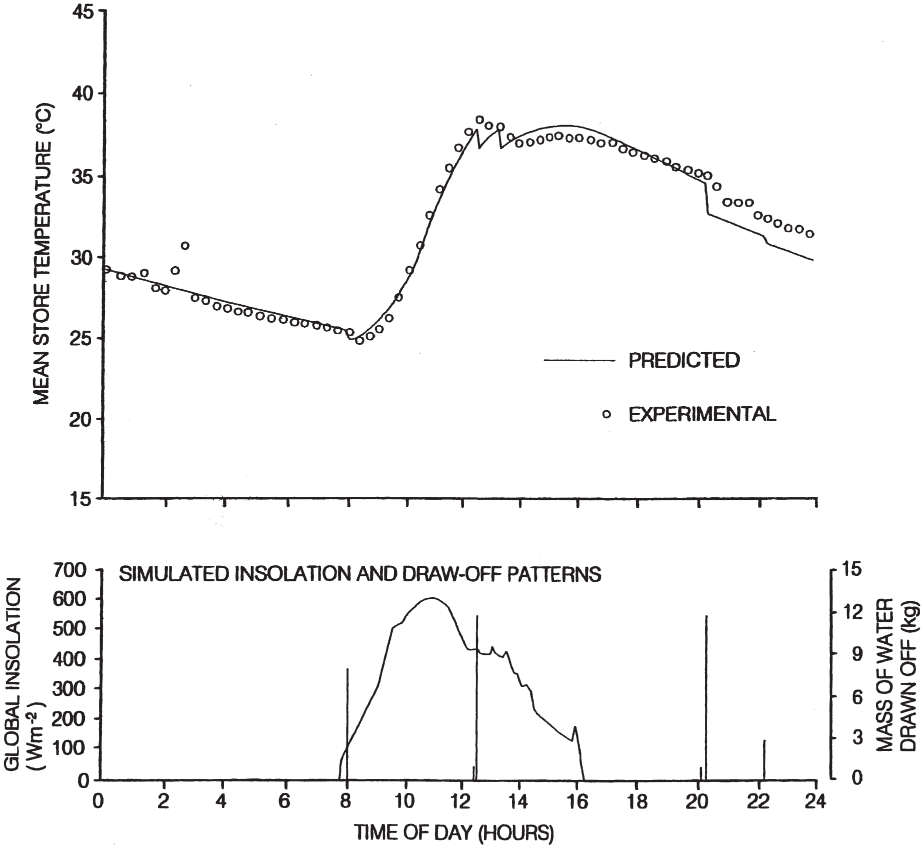

Fig. 5. Diurnal variations of experimental and predicted, via a rigorous simulation model, mean store temperature of a thermosyphon solar energy water heater.

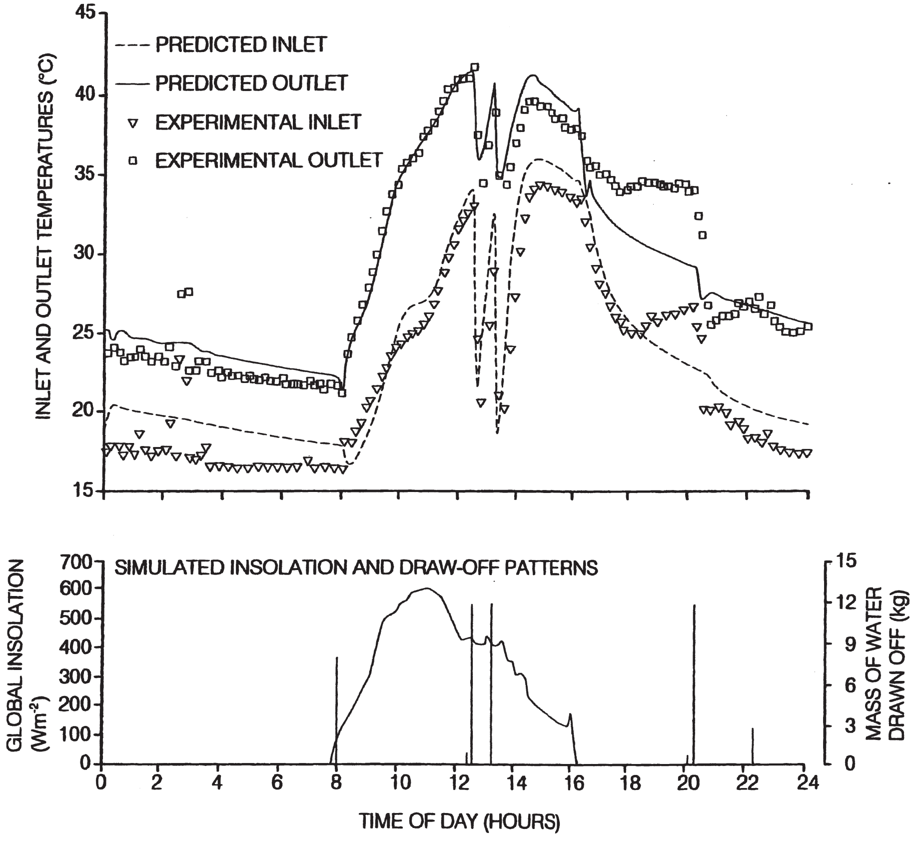

Fig. 6. Experimentally-observed inlet and outlet temperatures of a flat-plate collector in the flow circuit of a thermosyphon solar-energy water heater compared with those predicted from a rigorous simulation model.

correspond to within 7% of the measured values over the insolation period. Flow-rates were not measured during periods of reverse circulation as no signal was provided by the flowmeter below a positive flow of 10^{-4} kg s^{-1}.

The large fall-off and subsequent recovery in the measured flow rate immediately after water had been drawn off from the store was predicted correctly via the mathematical model. This reduced flow rate occurs as the drawn-off water is replenished by cooler water (at the mains supply temperature) introduced at the base of the store. This cooler fluid enters the collector and the resulting increase in fluid density in the risers and upriser pipes reduces the pressure differential between the fluid in these components and that of the store. A subsequent fall in the flow rate ensues. The reduced flow rate caused the fluid temperatures in the collector to increase; the fluid density falls and a higher rate is restored rapidly. The total thermal output due to the 7 draw-offs made during the day was measured as 4.27×10^{-6} J. The corre-

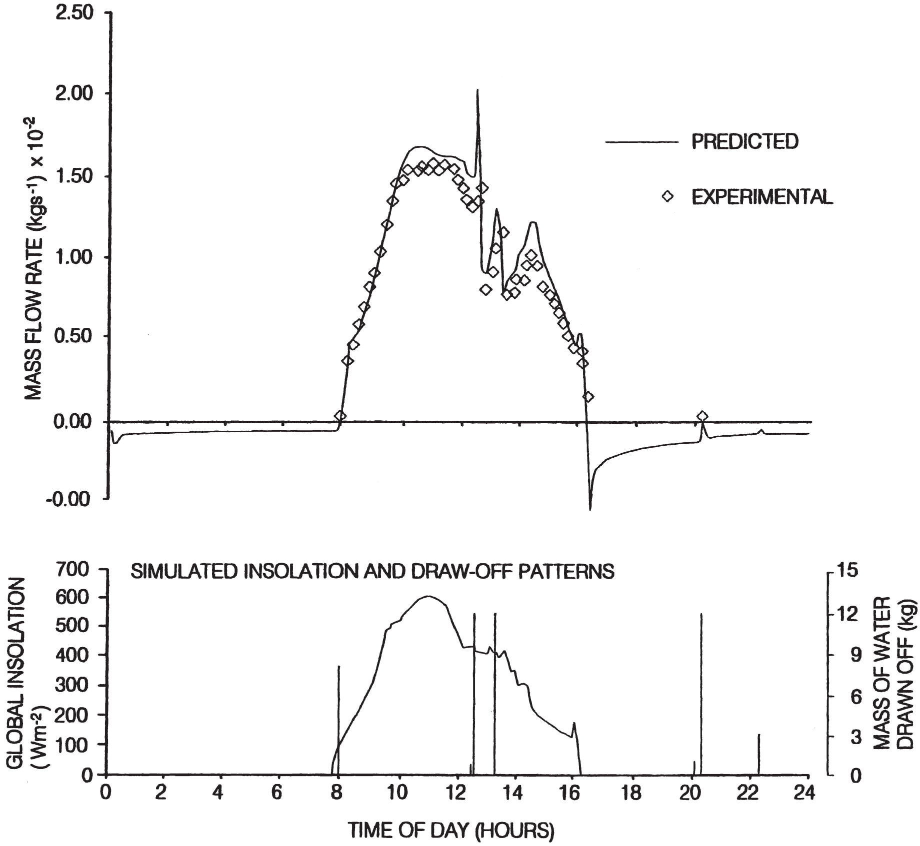

Fig. 7. Diurnal variation of experimentally-observed thermosyphonic mass flow rates and those predicted from a rigorous simulation model.

sponding predicted value was 4.54×10^{-6} J. This larger heat output can be attributed directly to the higher degree of stratification in the store predicted by this particular simulation model. Rigorous dynamic simulation models have been used for both direct [2] and indirect [9] thermosyphon solar water heaters, the latter have heat-exchangers in the store through which an anti-freeze solution is circulated to the collector. Detailed rigorous simulation models have been employed to determine the optimal concentration for an anti-freeze solution [8]. A dynamic simulation model with a three-dimensional representation of the hot water store has been employed to analyse the effect of tank geometry on thermal stratification [1].

5. Conclusions

The three generic approaches discussed for the analysis of thermosyphon solar energy water heaters are summarised in Table 1. The simplified approach described,

Table 1
Salient features of alternative generic approaches to the analysis of thermosyphon solar-energy water heaters

	Roles	Limitations	Advantages
Simplified models	• System sizing	• Only apply to the range of systems and operating conditions for which the simplifying assumptions are valid. • Often require experimentally-determined information only obtainable after the construction of the unit.	• Small input data set
Correlation performance characteristics	• Long-term performance prediction • Design charts • Economic optimisation	• Cannot be applied reliably to those systems for whose dimensions and climatic conditions, a correlation has not been determined.	• Small input data set • Wide-range of systems can be considered • Designer orientated
Rigorous simulation models	• Engineering optimisation • Establishing long-term performance • Determining validity of simplified models	• Require increased computing power • Large input data set	• Accurate transient simulation • All system configurations can be included

i.e. the ΔT model, is very quickly executed. Its application is limited by the need for the mean diurnal insolation intensity to exceed a, usually unknown, threshold level for a given system geometry. The principal generic shortcoming of such simplified models is that there is no prior way of determining the additional performance parameter, in this case the collector temperature differential ΔT, knowledge of which facilitates the simplification of the analysis, without performing either an experimental measurement or a rigorous simulation. However, for the particular case of the ΔT model, an analysis of the effects of changes in the ΔT value employed revealed that though the fluid mass flow rate is very sensitive to this factor, the final fluid temperature is much less affected.

The performance correlation described enables the long-term monitored performance of an indirectly heated thermosyphon solar-energy water-heater to be related, on a daily basis, to the thermal demands made on the system and the prevalent meteorological conditions. The correlation exhibits a high degree of linearity over a wide range of climatic conditions, and draw-off patterns.

Rigorous dynamic simulation models have required ‘expert’ users. However rapid and continuing advances in graphical user interfaces and pre- and post-processes together with the ready availability of computing capacity may render the bespoke

design of large-scale non-domestic thermosyphon solar water heaters practicable using such models as tools.

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